CHILLED WATER PLANT OPTIMIZATION BASED ON PART-LOAD COOLING TOWER PERFORMANCE

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ABSTRACT

Cooling towers are often used in air conditioning applications to reject heat into the atmosphere. For many people in the industry, the thermal performance of cooling towers is often taken for granted. With proper control of condenser water temperature from the cooling tower, up to a 27% increase in efficiency of a central chilled water plant can be realized.

The condenser water system for a chilled water plant usually consists of three major components: water chillers, cooling towers and circulating water pumps. The chilled water plant's overall efficiency and performance varies continuously throughout the day's operation depending on several parameters, one of which is the thermal performance of the cooling towers. The thermal performance of a cooling tower depends on the air and water flow rates, ambient wet bulb temperature and the entering and leaving water temperature. The objective of this research is to optimize the condenser water system to provide maximum plant efficiency during part-load conditions. To achieve this, a computer model was developed to assess the optimum power consumption of a central chilled water plant under various cooling loads and ambient temperature conditions by varying the leaving condenser water temperature. Simulations using the algorithms showed that it was able to predict the heat duty of a cooling tower within 2% of the manufacturer's rating data. With proper control of the cooling tower up to 27% increase in the efficiency of a central water plant was demonstrated.

INTRODUCTION

Two major components in a constant volume condenser water system for a central chilled water plant consist a chiller and cooling tower. These components are usually chosen using extreme conditions, such as maximum summer time temperatures and cooling loads. While these components are purchased to operate at 100% of their capability based on their required design conditions, nearly all hours of the equipment's operation throughout the year is considered off-design or at part-loads.

A relatively common control strategy for cooling towers during these part-load conditions is to maintain a fixed leaving condenser water temperature for all refrigeration load conditions. However, this does not provide the minimum amount of power being used by the central plant. During part-load operation of a water-cooled central plant, the capacity of the cooling tower should be controlled for effective operation of the refrigeration system to minimize the total plant's energy consumption.

Because of the increased use of microprocessor-based controllers the past few years, many new control strategies have been developed through the use of computer software for energy management systems that incorporate direct digital control systems. These new strategies can predict and adapt to various conditions to minimize the energy usage while maintaining optimum environmental conditions. Significant energy savings can be achieved in a central chilled water plant by optimizing the condenser water system.

The objective of this analysis is to:

1) Develop a mathematical model to predict a cooling tower's performance with varying wet bulb temperatures, tower air flow rate and range (temperature difference between the inlet and outlet water conditions).

2) Develop a mathematical model to predict a chiller's performance under varying refrigeration capacities and condenser inlet water temperature.

3) Integrate the models and the simulation of the condenser system to arrive at an optimized system.

4) Investigate the savings potential of an optimized condenser water system, sizing selection of cooling tower and optimum approach for various conditions.

COOLING TOWER MODEL

In many HVAC applications, wet cooling towers are used to reject heat into the atmosphere. An inappropriately sized or poorly operated condenser system can decrease the efficiency of the entire central plant. Operating chillers at higher-than-design condenser temperatures increases the enthalpy of the refrigerant exiting the compressor which in turn decreases the efficiency of the chiller.

Heat removal from the water in a cooling tower is accomplished by a transfer of sensible heat due to the temperature difference of the air and water, and by latent heat which is equivalent to the mass transfer resulting from the evaporation of water [1].

The cooling tower performance prediction used today is directly related to Merkel’s deduction. Merkel assumed the ratio of the overall sensible heat unit conductance to the mass transfer unit conductance is equal to one [2]. This assumption allows the overall process’s driving force to be based on the enthalpy difference. The deduction considers each water droplet being surrounded by a film of saturated air from which the sensible heat and mass is transmitted from the bulk hot water to the air stream. This method is well documented and will not be discussed in this paper

A computer algorithm was developed to calculate the cold water temperature, water leaving the tower, given the design parameters, wet bulb temperature and hot (entering) water temperature. The algorithm is not intended to replace the manufacturer’s data; however, it will be used to simulate the energy usage and tower performance for arbitrary loads and weather conditions. For this paper, a mechanical draft crossflow cooling tower has been chosen for development of the model using a finite difference method.

Because of horizontal and vertical variations, the tower must be divided into N pieces of small unit volume elements (Figure 1) to calculate the heat transfer between the water and the air.

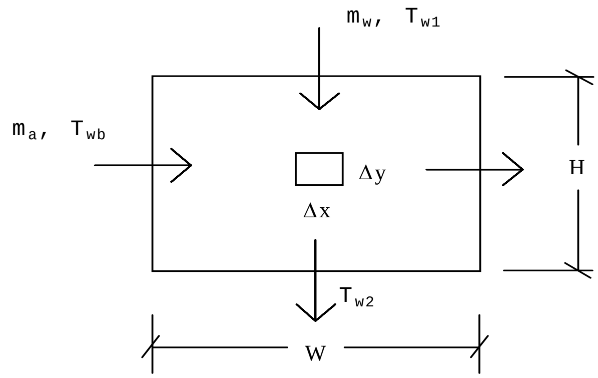

Figure 1 - Schematic of unit volume element for a crossflow cooling tower.

From Merkel's assumption the following relationship was developed

$$\frac{L\,c_{pt}\Delta x\Delta t}{W} = \frac{G\,\Delta h\,\Delta y}{H} = K\,a\,(h' - h)\,\Delta x\,\Delta y \qquad (1)$$

where:

c_p	=	Specific heat of water	[kJ/kg K]
L	=	Water flow rate of tower	[liter/hr]
Δt	=	Temperature fall of water in element	[C]
Δx	=	Width of differential element	[m]
W	=	Width of the element	[m]
G	=	Air flow rate of tower	[m^3/hr]
Δh	=	Enthalpy rise of air in the element	[kJ/kg]
H	=	Length of the element	[m]
Δy	=	Height of differential element	[m]
K	=	Overall mass transfer coefficient between saturated air and the main air stream	[kg/hr-m^2 (kg/kg)]
a	=	Area of water droplet interface	[m^2 /m^3]
h'	=	Enthalpy of saturated air surrounding the water droplets at the water temp. in the differential element	[kJ/kg]
h	=	Enthalpy of air at the wet bulb temp. in the element	[kJ/kg]

The L/G ratio only applies to any point in the tower if W = H, and the ratio of the number of vertical-to-horizontal differential elements will equal the ratio of the height-to-width if *dx* = *dy*. The calculations are simplified by considering incremental volumes that are geometrically similar in shape to the tower cross section [3] which yields

$$\frac{W}{\Delta x} = \frac{H}{\Delta y} = N \tag{2}$$

From Equation (1) and (2)

$$L c_p (t_i - t_o) = G (h_o - h_i) = \frac{K a V (h' - h_a)_{avg}}{N} \tag{3}$$

The temperature change of the water as it travels vertically through each element assuming that the specific heat c_p = 4.19 is

$$\Delta t = \frac{K a V}{4.19 * L * N} (h' - h_a)_{avg} = \frac{(NTU)}{N} (h' - h)_{avg} \tag{4}$$

and the change in the enthalpy of the air as it travels horizontally through each element is

$$\Delta h = (\frac{L}{G}) c_p \Delta t \tag{5}$$

From Equation (4) and (5), the outlet water temperature can be solved for by an iterative process with the unknowns for the equations being h_o, h_o' and t_o. However, h_o' is a function of t_o which leaves two equations and two unknowns.

From Webb and Villacres [3], the enthalpy of the entering air is a function of the wet bulb temperature which can be approximated by

$$h = 0.43 * t_{wb} + (1075 - 0.776 * t_{wb}) * (\frac{0.62198 * P_{sa}}{P_{atm} - P_{sa}}) + 7.69 \tag{6}$$

where,

t_{wb} = Air wet bulb temperature [C]
P_{sa} = Sat. vapor pressure @ temperature t_{wb} [N/m^2]
P_{atm} = Atmospheric air pressure [N/m^2]

The above equation is modified from the moist air enthalpy equation presented in Chapter 6 of the ASHRAE Handbook of Fundamentals [4] by substituting the wet bulb temperature in place of the dry bulb temperature. This will yield the enthalpy of saturated air. This approximation provides significant precision for this analysis, but a decreases in accuracy is expected as the ambient dry bulb temperature increases and the relative humidity is low (<30%).

The enthalpy of the saturated air at the bulk water temperature which can be determined by

$$h = 0.43 * t_{wb} + (1075 - 0.776 * t_{wb}) * \left(\frac{0.62198 * P_{sw}}{P_{atm} - P_{sw}}\right) + 7.69 \qquad (7)$$

where,

P_{sw} = Sat. vapor pressure at temperature t_w [N/m^2]

The saturated vapor pressures for the previous two equations are determined from the empirical equation developed by Hyland and Wexler [5].

A computer program was developed to perform this iteration until the value of heat gained or lost converges to a preset value for all elements. Input parameters to the computer program are

1)	Design Water Flow Rate	[liters/min.]
2)	Design Air Flow Rate	[m^3/min.]
3)	Design Wet Bulb Temperature	[C]
4)	Design Range	[C]
5)	Entering Hot Water	[C]
6)	NTU (From manufacturer)	

Once the input parameters are known and have been entered, the program is initialized. The program calculates the cold water temperature for varying air flow rates, ranges and wet bulb temperatures. From this output data, three bi-quadratic equations were developed to determine the cooling tower leaving water temperature for any given air flow rate, range or wet bulb condition. These curves will be used in the final integrated system model. The bi-quadratic equation will be in the format of

$$a + bx + cx^2 + dy + ey^2 + fxy$$

where the ECDW temperature will be determined by

$$ECDW = f_1(Wetbulb, Range) * f_2(Wetbulb, CFM) * f_3(Range, CFM)$$

WATER CHILLER MODEL

Most refrigeration systems used in air-conditioning today are vapor compression refrigeration systems. In vapor compression systems, the refrigerant is compressed to a higher pressure and temperature level after it has produced its refrigeration effect. The compressed refrigerant transfers its heat to the sink (condenser water) and is condensed into the liquid phase. The liquid refrigerant is then throttled to a low pressure, low temperature vapor to produce a refrigeration effect during condensing.

The performance of a water chiller is dependent on the following parameters:

1)	Leaving chilled water temperature	LCHWT
2)	Entering condenser water temperature	ECDWT
3)	Chilled water flow rate	CHW
4)	Condenser water flow rate	CDW
5)	Chiller capacity	

Assuming that the CHW and CDW are constant volume by design and the LCHWT is held constant, a significant increase in efficiency can be realized by reducing the ECDWT. For a simple refrigeration cycle shown in Figure 2, the design coefficient of performance (COP_D) is described by the enthalpy points 1,2,3 and 4.

$$COP_D = \frac{h_1 - h_4}{h_2 - h_1} \quad (8)$$

Operating a chiller at reduced condenser temperatures decreases the enthalpy of the refrigerant exiting the compressor (h_2'). The change in the efficiency for the simple refrigeration cycle is:

$$COP = \frac{h_1 - h'_4}{h'_2 - h_1} > COP_D \quad (9)$$

To calculate the performance of a water chiller, a model was developed that utilized manufacturer’s performance data for two different types of machines -- a centrifugal and screw chiller.

For simplification, a nominal 1,406 KW machine was selected for each type with the following design criteria and assumptions:

1) 29.4 to 35°C on condenser water
2) 11.1 to 5.6°C on chilled water
3) 75.7 liters/sec. condenser water flow rate
4) Leaving chilled water temperature is constant
5) Both chilled and condenser water flow rates are constant

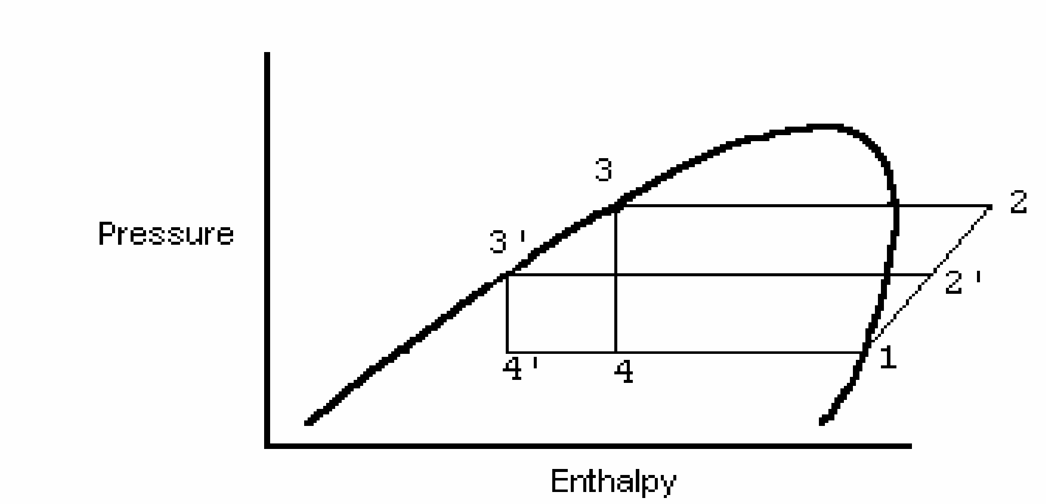

Figure 2 - Pressure-enthalpy diagram for a simple refrigeration cycle.

The electrical input to the chiller is calculated by

$$Elec = Eff_d * Cap_d * PLR * Eff_{PLR} * Eff_{CW} * Frac \qquad (10)$$

where:

Elec	=	Input power of chiller	[kW]
Eff_d	=	Design efficiency of chiller	[kW/KW]
Cap	=	Design capacity of chiller	[KW]
PLR	=	Chiller's part-load ratio	
Eff_{PLR}	=	Efficiency part-load correction factor (quadratic equation)	
Eff_{cw}	=	Efficiency condenser water correction factor (quadratic equation)	
Frac	=	Fraction of the hour the chiller operates below minimum load	

Manufacturer's data was obtained for a nominal 1,406 KW chiller with the design parameters stated earlier. It is also evident, from comparing the performance curves, that the screw chiller has better performance at part-loads than the centrifugal chiller. However, the screw machine also has a better performance at lower entering condenser water temperatures than a centrifugal machine.

INTEGRATION OF MODELS

To determine the optimum approach of the condenser water system and therefore the optimum system input power, both the cooling tower and chiller models must be combined. To accomplish this task, the final model will integrate the inter-relationship of the variations in the ambient wet bulb temperature and chiller loading.

Certain assumptions were made to provide an accurate integrated model. First, it was assumed that the cooling tower fan has infinite variable air flow rate over the range of 100% to 45%. A minimum air flow rate was set so that the motor and/or gearbox would not be damaged. The value for this analysis was placed at 45%; however, lower values can possibly be used.

Second, the chiller requires a differential pressure between the evaporator and condenser to operate. A differential pressure is required to force the refrigerant to where it can effectively do its job. It was assumed for this analysis that the minimum condenser water temperature entering the chiller is 15 ^{o}C.

From the integrated model, two simulations were made. The first simulation was generic in the sense that it determined the optimum input power for the chiller and cooling tower for a given load range at every wet bulb temperature, ranging from 6.7 to 25.5 ^{o}C. This optimum input power was then compared to two different condenser water control strategies:

1) Maintaining a fixed condenser water temperature (26.7 ^{o}C)
2) Maintaining the coldest condenser water temperature possible.

The above control strategies were modeled as a single speed cooling tower fan that cycled to maintain a condenser water setpoint. Once the condenser water temperature fell below the leaving water setpoint (or minimum) temperature, the fraction of the hour the fan operates is determined by

$$(R * T_{LS} - T_{LS}) \big/ (R * T_{LS} - T_{LM}) \qquad (11)$$

where

R = Range of the cooling tower

T_{LS} = Leaving condenser water temperature setpoint or minimum

T_{LM} = Leaving condenser water temperature when tower fans are operating

The second simulation uses data from an hourly energy analysis program for two distinctly different facilities. Again the optimum power input was compared to the two different control strategies presented earlier. Also, an economic analysis was conducted to determine which size cooling tower (design approach) should be used for each type of facility.

OPTIMUM TOWER CONTROL

As shown in Figure 2 and from the manufacturer's performance data, a reduction in the inlet condenser water temperature typically increases the efficiency of the chiller. This increase in chiller efficiency comes at the expense of tower fan energy. Figures 3 and 4 below show the results of the integrated model at 21.1 ^{o}C ambient wet bulb and 80% of full load (1,125 KW) for both types of chillers. The chiller KW, cooling tower fan KW and total system KW are plotted with different entering condenser water temperatures.

The optimum ECDW temperature for Figure 3, where the system KW is minimum, is 26.1 ^{o}C. The corresponding approach and percent of the rated air flow is 4.99 ^{o}C. and 79% respectively. One degree below the optimum ECDW temperature, the chiller consumes 1.4% less power but the tower fan consumes 73.1% more power. One degree above the optimum ECDW temperature, the tower fan consumes 33% less power but the chiller consumes 2.3% more power. The net effect is that the total input power for the system increases 1.7% and 0.67% higher than the optimize system KW respective.

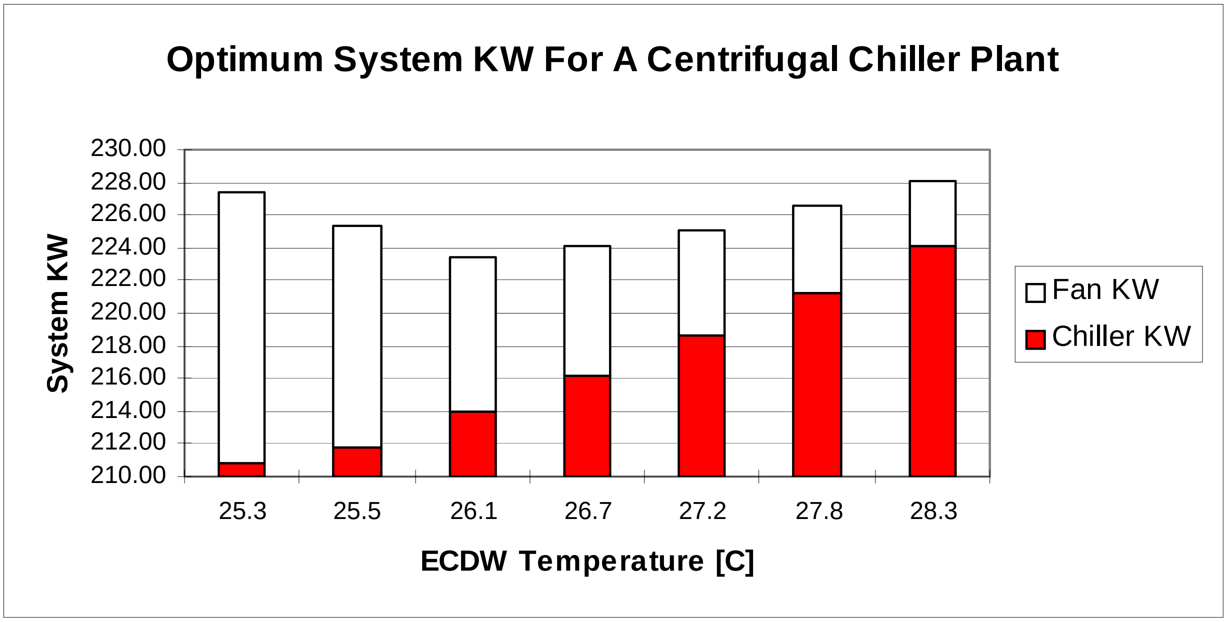

Figure 3 - Integrated system model for a centrifugal chiller and 3.9^{o} design approach tower at 80% load and 21.1 ^{o}C. wet bulb temperature.

Similar values can be seen with the screw chiller presented in Figure 4. The main difference between the two types of chillers is that the percent air flow rate for the optimum system in the screw chiller is higher than that of the centrifugal chiller; yielding a lower optimum, ECDW temperature for the screw chiller.

A classical control method used in industry is to maintain a constant ECDW temperature for all loads and wet bulb temperature variation. This method minimizes fan energy consumption for various loads and wet bulb temperatures, but fixes the efficiency correction factor for the ECDW temperature for all operating periods.

Another control strategy is to produce the coldest condenser water temperature possible. This method primarily fixes the cooling tower fan energy consumption while trying to reduce the chiller consumption. Figures 5 and 6 compares the optimum control strategy to the previous strategies stated above for both types of chillers.

The full-load input power to the centrifugal and screw chillers are 227.2 KW and 284 KW respectively with a full-load input power of 16.57 KW from a 3.9 ^{o}C design approach cooling tower.

It can be seen in Figures 5 and 6 that at full load conditions at 25.5 ^{o}C wet bulb temperature(design conditions), the total

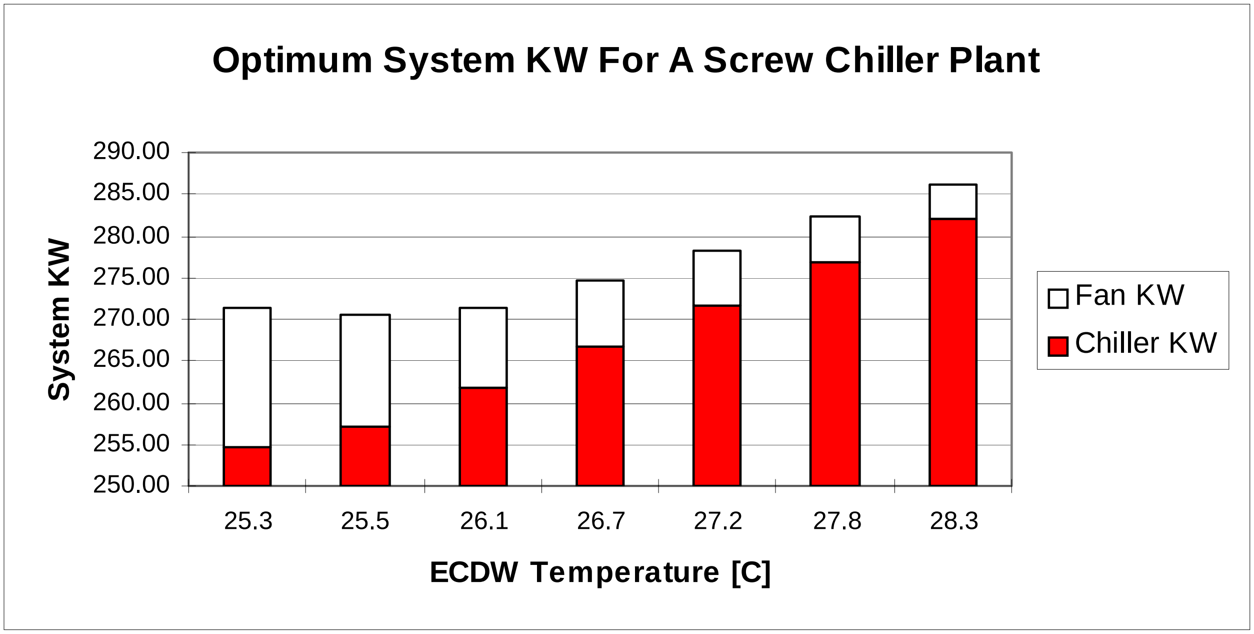

Figure 4 - Integrated system model for a screw chiller and 3.9^{o} design approach tower at 80% load and 21.1 ^{o}C. wet bulb temperature.

system input power for the optimum condition is less than the design system input power. For this condition, the percent of the design air flow rate that yielded the minimum input power for the centrifugal and screw systems is 85% and 95% respectively.

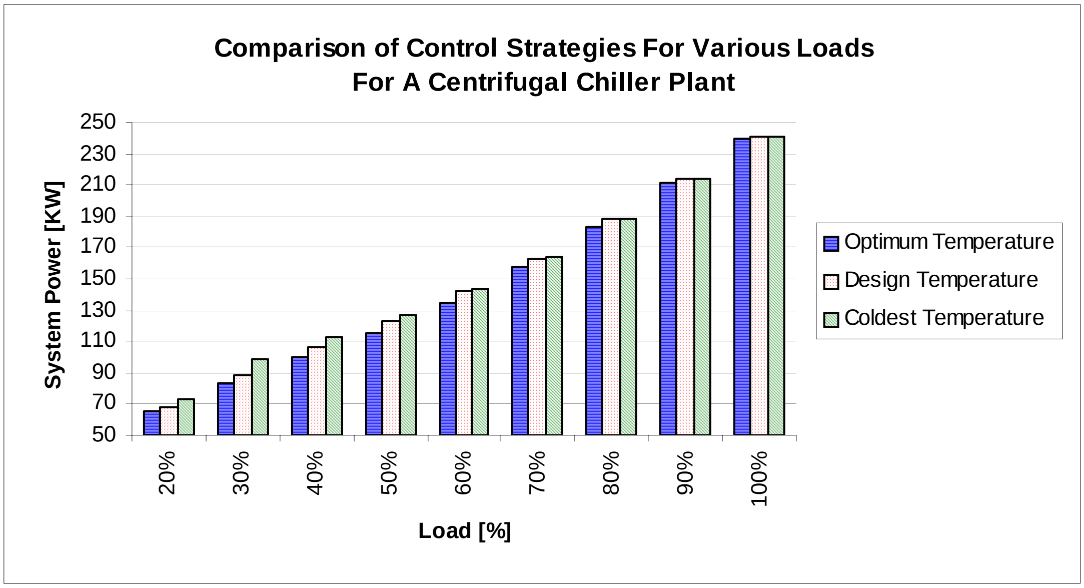

Figure 5 -Comparison of different control strategies for a centrifugal chiller plant at various loads and ambient wet bulb temperatures.

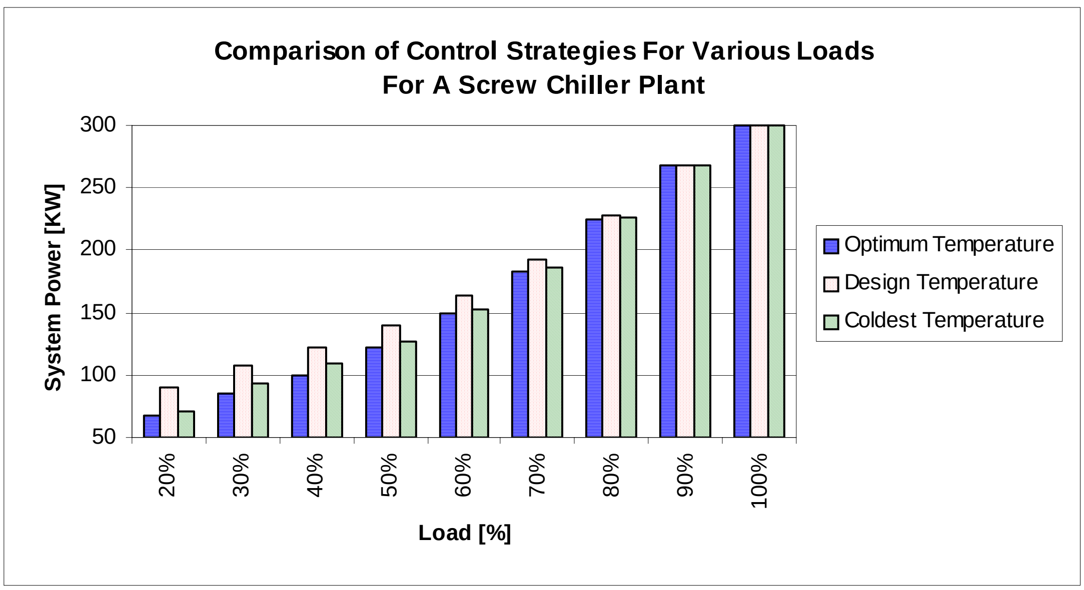

Figure 6 -Comparison of different control strategies for a screw chiller plant at various loads and ambient wet bulb temperatures.

It can also be seen when examining Figure 6, that maximum savings available for the screw chiller system is when the optimum method is compared against the fixed temperature control method. For the centrifugal chiller system (Figure 5), the maximum available savings occur when it is compared to the coldest possible control method. For both systems, it is noticed that the percent saving increases with a decrease in load and ambient wet bulb temperature. This presents a question as to whether or not a different size cooling tower may be a better economic advantage over the life of the cooling tower.

SIMULATED RESULTS

To determine realistic savings potential for an optimized condenser system and to investigate the most economical cooling tower size, a simulation must be performed using actual data for a specific facility. Two computer models, a resort facility and an elementary school, were developed using a hourly analysis program. These two types of facilities were chosen for their extreme differences in operating hours and load profiles.

A hourly load profile for a central plant was developed for each facility using Tampa, Florida design temperatures from ASHRAE [6] and hourly TMY weather data. Also, performance curves for three different cooling towers were developed for design approaches of 1.67^{o}, 2.8^{o} and 5.5^{o}. The model was then simulated to determine the annual savings expected using the optimum control strategy compared to the more common control strategies for each size tower and facility type.

From the first simulation presented earlier, it was observed that the major savings potential occurred when the chiller operated at a low load compared to a lower wet bulb temperature. Figure 5 below compares the equipment’s hours of operation for different load ranges and it could be assumed that the maximum percent savings in the system input KW will be greater for the elementary school due to its high number of low load operating hours compared to the resort facility. This assumption is validated in Tables 1 through 4.

To determine which size cooling tower would provide the best economic effectiveness for each type of facility, a “net present worth” economic analysis was performed. This method is defined to be the difference between the present worth of the project revenues and project costs. The “net present worth” was determined by the following

$$NPW = \sum_{t=1}^{t=N} \frac{c(t)}{(1+i)^{t}} - initial$$

where

NPW = Net Present Worth
N = Economic life
i = Annual interest rate
c = Operating cost of chiller and cooling tower
initial = Initial capital outlay

Approach	Elementary School		Resort	
	Opt./Fixed	Opt./Coldest	Opt./Fixed	Opt./Coldest
1.67 C	4.36%	26.52%	6.66%	12.97%
2.8 C	3.68%	25.79%	5.90%	12.13%
3.9 C	2.38%	17.05%	3.50%	6.55%
5.5 C	1.58%	10.21%	3.95%	2.67%

Table 1 - Percent savings of the different control strategies for a centrifugal chiller with different cooling tower approaches.

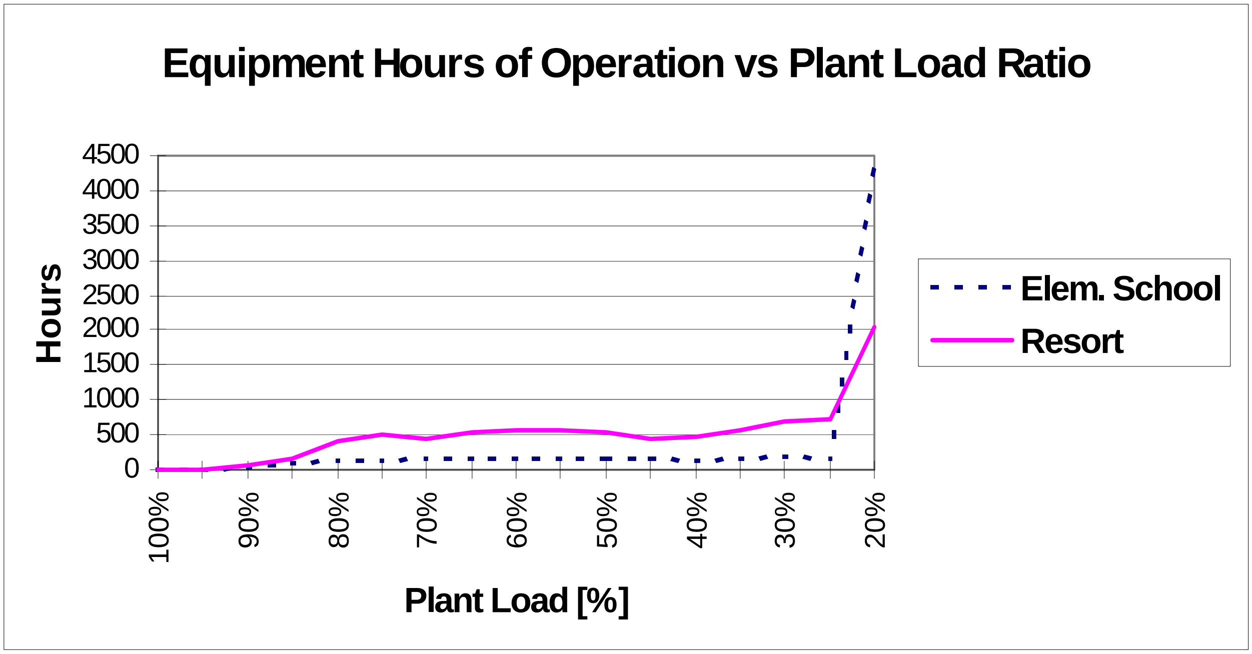

Figure 7 - Comparing facilities' hours of operation at various load ranges.

Approach	Elementary School		Resort	
	Opt./Fixed	Opt./Coldest	Opt./Fixed	Opt./Coldest
1.67 C	11.09%	23.42%	13.65%	10.73%
2.8 C	9.14%	22.04%	11.39%	9.00%
3.9 C	7.31%	13.72%	8.44%	3.93%
5.5 C	6.01%	7.65%	6.53%	1.16%

Table 2 - Percent savings of the different control strategies for a screw chiller w/ different cooling tower approaches.

ANNUAL SYSTEM CONSUMPTION FOR A RESORT FACILITY						
Approach	Centrifugal Chiller			Screw Chiller		
	Optimum	Design	Coldest	Optimum	Design	Coldest
1.67 ^{0}C	946,244	1,013,735	1,087,320	1,042,568	1,207,316	1,167,907
2.8 ^{0}C	961,215	1,021,478	1,093,878	1,077,444	1,215,964	1,184,012
3.9 ^{0}C	965,979	1,001,001	1,033,697	1,099,318	1,200,628	1,144,277
5.5 ^{0}C	977,567	1,017,807	1,004,397	1,121,695	1,200,095	1,134,812

Table 3 - Annual system consumption for different control strategies for the various cooling tower design approaches for the resort facility.

ANNUAL SYSTEM CONSUMPTION FOR AN ELEMENTARY SCHOOL						
Approach	Centrifugal Chiller			Screw Chiller		
	Optimum	Design	Coldest	Optimum	Design	Coldest
1.67 ^{0}C	346,075	358,281	466,346	388,333	432,957	502,650
2.8 ^{0}C	351,238	360,820	468,339	399,611	435,731	507,831
3.9 ^{0}C	346,108	352,096	414,356	399,657	428,725	460,406
5.5 ^{0}C	343,655	347,493	380,886	402,730	426,725	434,283

Table 4 - Annual system consumption for different control strategies for the various cooling tower design approaches for the elementary school facility.

The most economically effective project will have the smallest net present worth. For this economic analysis the following assumptions were made

1) economic life of project is 20 years [7]
2) no project revenue was used
3) capital outlay for each cooling tower was provided by a cooling tower manufacturer

4) a virtual rate of $0.10/KWH is used with no inflation over the life of the analysis
5) an annual interest rate of 8%

Approach	Tower Cost	CENTRIFUGAL CHILLER		SCREW CHILLER	
		Elementary	Resort	Elementary	Resort
1.67 ^{0}C	$66,500	$369,811	$904,056	$407,216	$958,884
2.8 ^{0}C	$49,500	$357,115	$900,307	$399,931	$1,003,186
3.9 ^{0}C	$39,450	$343,672	$894,473	$391,070	$1,012,498
5.5 ^{0}C	$35,100	$337,805	$900,381	$390,094	$1,027,954

Table 5 - Results of the net present worth analysis for each type of chiller and load profile.

As evident in Table 5 for both the centrifugal and screw chillers for an elementary school, the 5.5^{0} design approach cooling tower yielded the most economically attractive scenario. For the resort facility, the 3.9^{0} approach cooling tower provided the best economics for the centrifugal chiller plant; however, for the screw chiller the 1.67^{0} approach tower was selected.

The major factor in the net present worth analysis for the centrifugal chiller is the initial capital cost. It can be seen in Tables 3 and 4 that the annual consumption for the optimum control strategy varied very little between the different cooling towers. However, the opposite is true of the screw chiller. In Table 3, the difference in the annual consumption of the various cooling towers is much more pronounced thus reducing the effect of the initial capital expense.

Since a screw chiller has better performance characteristics at lower loads and ECDW temperature then a centrifugal chiller, it stands to reason that a lower design approach cooling tower would provide a more attractive economic outlook. The economic analysis presented in this paper shows that the above statement is true, but the amount of EFLHR is a major governing influence in how large the cooling tower should be.

OPTIMUM CONTROL OF COOLING TOWER APPROACH

A nominal 1406 KW chiller and a 3.9 ^{0}C design approach cooling tower were modeled. From the simulation performed on the model, the approach that provides the optimum system input power was calculated. Figures 8 and 9 shows the optimum approach for both types of chillers versus various loads for different wet bulb temperatures.

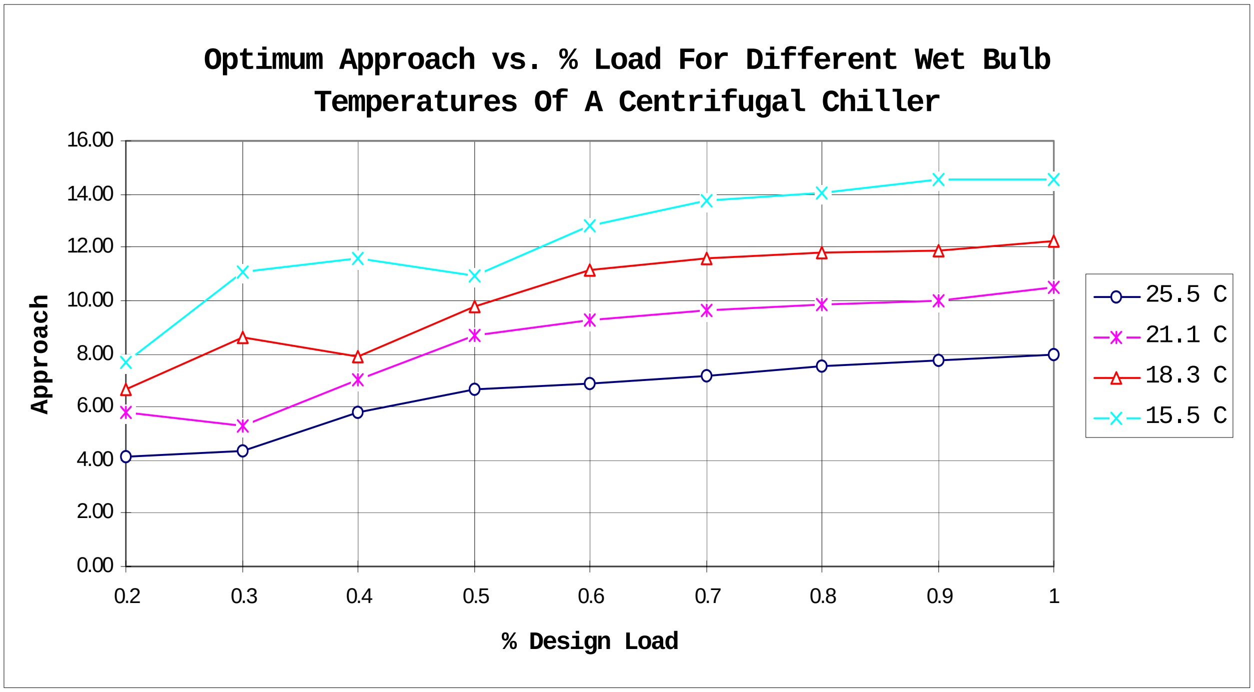

Figure 8 - Optimum approach for a centrifugal chiller and 3.9^{0} design approach cooling tower.

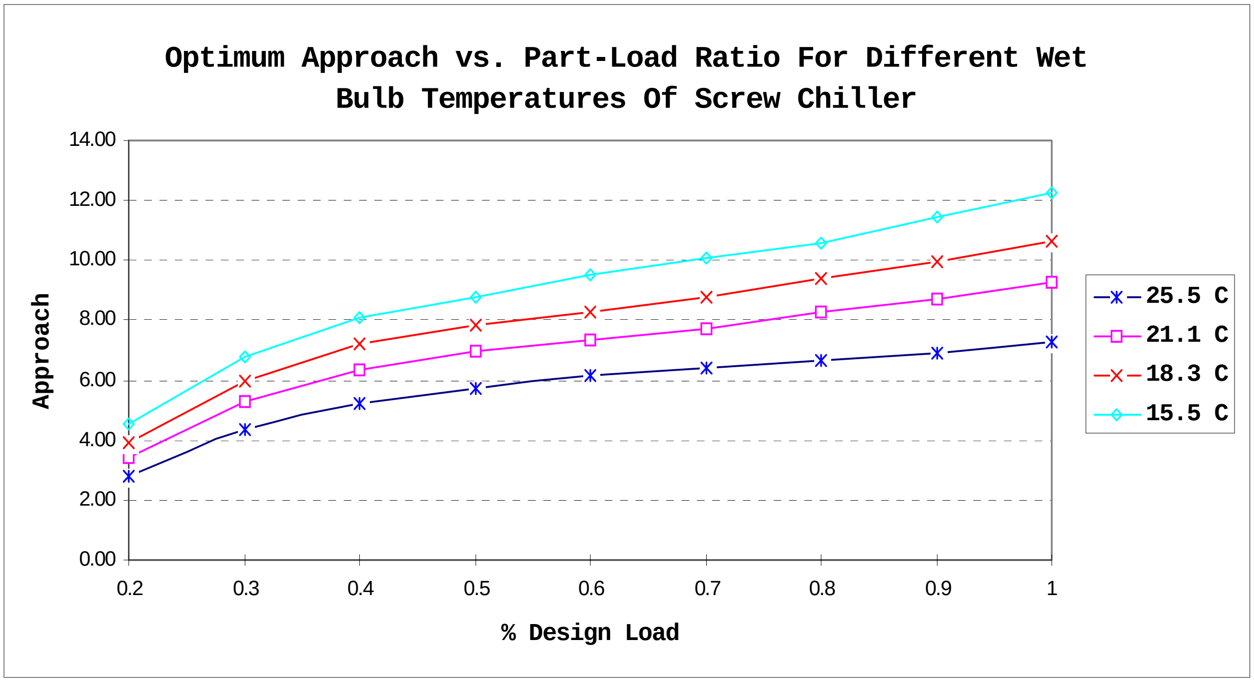

Figure 9 - Optimum approach for a screw chiller and 3.9^0 design approach cooling tower

The optimum cooling tower approach that would minimize the chiller and cooling tower fan input power is directly related to on the performance characteristics of the cooling tower and chiller. Therefore, the data presented in this paper will have merit for chillers and cooling towers that exhibit the performance characteristics similar to what is modeled.

With the advancement in microprocessor technology, the Energy Management Control System (EMCS) can more effectively optimize the operations of most HVAC systems thus minimizing the utility cost of a facility. For facilities that utilize a EMCS, a simple algorithm could be used to yield substantial energy savings.

The simplest algorithm would be to provide a multi-variable equation that would calculate the best curve fit for the data produced by the computer simulation. This could be accomplished by the bi-quadratic equation

$$ECDW_{OP} = [a + b * PLR + c * PLR^2 + d * WB + e * WB^2 + f * PLR * WB] + WB$$

where

$ECDW_{OP}$ = Optimum entering condenser water temperature
WB = Current ambient wet bulb temperature

PLR = Current part-load ratio

Where this is the simplest algorithm to input, determining the coefficient of this equation would be exceedingly more difficult. However with a microprocessor-based control system, a set of algorithms such as the one developed for this paper could be programmed into the EMCS to determine the optimum condenser water temperature setpoint for any conditions.

CONCLUSIONS

Manufacturer’s data is used to develop mathematical performance models for chillers and cooling towers to analyze the optimum total system input power. The results show that up to 27% savings can be achieved if proper control of the condenser water system is implemented. A simple control algorithm was developed that would provide optimum condenser water system while continuing to satisfy the current load on the facility.

The model developed for this paper was for an infinitely variable air flow rate over a given range to determine the optimum system input power of a chiller plant. However, in many instances two-speed motors are used on cooling tower fans instead of adjustable speed drives. The algorithms developed for this paper could easily be modified to develop the optimum control strategy for a two-speed fan.

To obtain the greatest possible energy savings through optimization, each system in a chilled water plant must be monitored and controlled. With proper monitoring of specific points in a chilled water plant, the optimum ECDW temperature can be modified for performance changes in equipment overtime.

Further studies are recommended in condenser water system optimization that consider variable water flow rates in place of and in combination with variable air flow rates in a cooling tower. Also, it is recommended to analyze a central plant that has several different capacity chillers and cooling towers.

REFERENCES

[1] Baker, D.R. and Shryock, H.A., A Comprehensive Approach to the Analysis of Cooling Tower Performance, Journal of Heat Transfer, August, 1961.

[2] Anon., Chapter 36, “Cooling Towers” in ASHRAE inc., Handbook HVAC Systems and Equipment - 1996, ASHRAE, Atlanta, GA, 1996.

[3] Webb, R.L. and Villacres, A., Algorithms for Performance Simulations of Cooling Towers, Evaporative Condensers, and Fluid Coolers, ASHRAE Trans., Vol. 90, Part 2, 1984.

[4] Anon., Chapter 6, “Psychometrics” in ASHRAE inc., Handbook of Fundamentals - 1989, ASHRAE, Atlanta, GA, 1989.

[5] Hyland, R.W. and Wexler A., Formulations for the Thermodynamic Properties of the Saturated Phase of H_2O From 173.15 K to 473.15 K, ASHRAE Trans., Vol. 89, Part 2A, 1983.

[6] McQuiston, F.C. and Spitler, J.D., Cooling and Heating Load Calculation Manual, ASHRAE inc., Atlanta, GA, 1994.

[7] Anon., Chapter 33, “Owning and Operating Cost” in ASHRAE inc., Handbook Application - 1991, ASHRAE, Atlanta, GA, 1991.